# PATENT SPECIFICATION

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COMPLETE SPECIFICATION

## Improvements in Centrifugal Compressors for Supercharging Internal Combustion Engines

We, Dr. Ing. Alessandro Baj, Via Monte Amiata 3, Milan, Italy, of Italian Nationality, and Fabbrica Automobili Isotta Fraschini S.A., Via Monterosa 79, Milan, Italy, an Italian Company, do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

This invention relates to centrifugal compressors for supercharging internal combustion engines.

It is known that, in order to obtain the best adiabatic efficiencies and the highest compression ratios from centrifugal compressors, the air accelerated by the compressor should strike against no impediments in its passage through the intake leading to the fan and between the fan and the diffusing blades. Experiment has also shown that changes in speed should be gradual and congruent. The higher the pressure-ratio of a fan, and therefore the higher the peripheral speed, the more important is it that correct and regular flow of the air stream is attained. In modern compressors for supercharging aircraft engines, in which the speed of flow of the fluid, both in the conduits and in the fan, has attained the highest values recorded in the art, it is of the utmost importance to prevent the fluid vein from striking against anything.

The present improvements have the object of impressing on the fluid vein a motion approaching very nearly theoretical flow conditions and ensuring that the fluid does not strike against any impediment on entering the fan.

According to the invention, a centrifugal compressor includes a fan adapted for axial intake of fluid and guiding elements at the intake to the fan, the guiding elements, or parts thereof, each being angularly adjustable about an axis located in a plane extending transversely of the fan axis, the arrangement being such that on appropriate adjustment of the guiding elements, or parts thereof, the fluid passing to the fan is guided to flow axially of the fan and also rotationally at linear speeds which immediately before the fan entrance and at different radii from the fan axis are substantially equal to the linear speeds at the corresponding radii of the fluid in the fan.

The guiding elements may include directional blading curved helicoidally and receiving the fluid from a helical conduit which imparts to the fluid a rotational component corresponding approximately to the mean average value of the rotational speed component of the fluid in the fan.

In practice, it is found that shockless conditions of inflow approximating to theoretical are obtained only for given values of the volume of flow, the delivery pressure and the rotational speed. When these factors vary, there should be a change in the inclination at different radii of the exit edge of the directional blading. This change can be obtained by making the outflow portion of said blading more or less inclinable. Such a change may be effected by a servo-motor controlled by the supercharging pressure feeding the engine.

A device used in many compressors for radial engines consists in using an intake spiral in front of the fan, having simple radial blading. This device has the drawback of causing a free vortex wherein the highest linear speeds are towards the centre (in accordance with the hyperbolic law) thus giving flow conditions which are the reverse of the real variation of linear velocities at different radii of the fan (namely, linear speed increasing with the radius).

The invention will now be described by way of example with reference to the accompanying diagrammatic drawings, in which:—

Fig. 1 is a sectional elevation of a compressor with directional blading having an end portion movable, and Fig. 2 is a development of some of the directional and fan blading.

Fig. 3 is a sectional elevation of a compressor with movable directional blading, and Fig. 4 is a corresponding developed view similar to Fig. 2.

Fig. 5 is another similar developed view illustrating an alternative, with

sectioned fan.

Fig. 6 is a sectioned elevation, drawn to a larger scale than the preceding views, of the preferred embodiment of a compressor with variable directional blading, and Fig. 7 is a section on line 7—7 of Fig. 6.

Fig. 8 is a velocity diagram.

Referring to Figs. 1 and 2, the fan or impeller G is of the straight radial type, and its vanes 10 (see Fig. 2) are axially straight. A shroud of convenient shape forms the spiral delivery chamber C. The intake P is formed by a spiral cowl having guiding elements constituted as helical directional blades 11 for the fluid. Each of these blades is constituted by a rear part 12 fixed to the cowl and an angularly adjustable front part 13. The part 13 of each blade is adjustable about an axis located in a plane extending transversely of the fan axis and all the parts 13 are adjustable simultaneously under the control of a lever 14.

In this manner, on appropriate adjustment of the parts 13, the fluid stream lines can be directed to assume a motion approaching very nearly to the motion of a fluid under theoretical conditions of flow and such that impact by the fluid at the entrance of the fan, with consequent formation of vortices and eddies, is avoided.

Moreover, the fluid stream lines radially of the fan are directed in the direction of that velocity which is the resultant of the peripheral velocity and the longitudinal (intake) velocity, i.e. the fluid as it leaves the parts 13 and immediately before the fan entrance has, besides an axial (longitudinal) motion, also a rotational motion with linear velocities at different points radially from the fan axis equal to the linear velocities at the corresponding points in the fan. The helical shape of the blading 11, due to the particular spiral, or snail volute, shape of the cowl, imparts to the incoming fluid a rotational component corresponding very nearly to the mean average value of the rotational component in the fan.

The alternative shown in Figs. 3 and 4 obtains the same result by bending the vanes 10*a* of the fan G forward in the direction of motion towards their intake, or fluid entrance, edges. The directional action of the blading 11*a* for the incoming fluid is achieved in this case by having the blades themselves angularly adjustable, so that they may be adjusted to the desired angle of entrance.

The cowl P in this alternative is not spiral form, this being unnecessary on account of the mobility of the rear part also of the blades 11*a*, which is pivotally mounted at their roots in a hub 15. This hub is curved ogivally to ensure that the fluid may pass around it without incurring losses.

In the embodiment shown in Fig. 5, the fan G has vanes 10*c* bent as in the case of Figs. 3 and 4, whilst the directional blading 11*c* has a double order of blades 16*c* and 17*c*. One or both orders comprise movable blades. When the blades of both orders are movable, they are adjusted at different intake angles selected so as to secure gradual change in the direction of flow. The blades may be controlled either by a single controlling member or by one member for each order of blades.

The principles upon which the invention is based may be carried out by providing a fan according to any of the previously described embodiments in combination with any of the described directional blading arrangements, as may be determined by various requirements due to particular technical conditions.

Likewise the control of the movable blades or blade parts may be effected in any of various manners. Figs. 6 and 7 show a preferred embodiment of means for effecting such control.

Referring to Figs. 6 and 7, the blades 11 each have a pivot pin 20 to which a small pinion 21 is fixed. Each pinion 21 meshes with a circular rack of gear teeth 22 cut in a crown 23 which is turnable in a seat on a ring 24 which is fitted into the intake chamber P of the compressor.

The crown 23 is controlled positionally by a pinion 25 which can be rotated from outside the compressor by a lever 26 operatively connected or not to means controlling the running conditions of the supercharger (intake pressure or outlet pressure, speed, outer atmospheric pressure, functional characteristics of the engine etc.).

A compressor made according to the principles underlying the present invention has improved operative features due to the dynamic conditions under which the fluid operates.

In the particular case of air-craft superchargers, the volume of air per second is proportional to the number of revolutions of the engine. Where a compressor is driven by the engine through a speed-multiplying gear with fixed ratio, and no reductions are made in the channel sections before or after the compressor, for each speed in revolutions per minute there corresponds a certain value of the air feed pressure. If one assumes working conditions at the altitudes for which the normal operations of the engine are calculated, the maximum engine output is obtained under such conditions and

with the highest pressure of air feed. If however the output has to be varied without varying the rotational speed, the feed pressure should be reduced, as stated above, by reducing the section of flow either before or after the fan. As regards the compressor, this means an increase in delivery and a reduction in the volume flow, which condition causes the operation of the compressor under conditions represented by another point of its characteristic curve and involves a change in adiabatic efficiency and in power input.

On the other hand, in use of a compressor provided with movable directional blading according to the present invention, it is possible to change factors of the theoretical water head pressure of the fan while keeping the rotational speed constant so that the new pressure head and volume flow correspond to a lesser power input obtained by throttling.

On analysing Eulers formula giving the theoretical head $H_t$ it is found that

$$H_t = \frac{U_2 C_{2u} - U_1 C_{1u}}{g}$$

If as first approximation one considers a radial outflow, one obtains

$$H_t = \frac{U_2^2 - U_1 C_{1u}}{g}$$

In these formulæ: —

$U$ = the peripheral speed;

$C_u$ = the peripheral component of absolute velocity.

$H_t$ = pressure measured in the fluid height.

$U_2$ = peripheral speed of the outside diameter of the revolving part.

$U_1$ = peripheral speed of the diameter of the revolving part at the entrance.

$C_{2u}$ = Absolute exit speed component according to the exit peripheral speed.

$C_{1u}$ = Tangential component of the absolute entrance speed.

We can vary the term $C_{1u}$ by varying the intensity in the direction of the absolute inlet velocity $C_1$ as shown in the velocity diagram, Fig. 8.

In such a case $H_t$ will vary and will decrease for any increment of $C_1$ and for any increment in the inclination of the blades (reduction of angle).

In the present case these quantities are bound one to another in so far that, on increasing the inclination of the blades, the free area of passage between the blades decreases.

These considerations of the triangles in the diagram show also that the integral value of the changes in the momentum (quantity of motion) impressed on the fluid depends on the inclination of the directional blades. Now the intake condition shows that a part of the energy of the fluid is returned to the fan precisely because it is found in this form. It will be therefore possible to obtain a decrease in the useful water head $H_t$ with a final energy balance-sheet better than when obtained by throttling.

In the case of radial blading with intake edge bent forwards according to the direction of the theoretical triangles, it is important for avoidance of shock to obtain an air-flow with streamlines strictly axial at inflow. In such a case the admission conduit has radial blading; and, when the compressor has to operate in varying conditions the blading is angularly adjusted.

In practice, particulars of construction may be varied without departing from the scope of the invention claimed.

Having now particularly described and ascertained the nature of our said invention and in what manner the same is to be performed, we declare that what we claim is: —

1. A centrifugal compressor for supercharging internal combustion engines, including a fan adapted for axial intake of fluid and guiding elements at the intake to the fan, the guiding elements, or parts thereof, each being angularly adjustable about an axis located in a plane extending transversely of the fan axis, the arrangement being such that on appropriate adjustment of the guiding elements, or parts thereof, the fluid passing to the fan is guided to flow axially of the fan and also rotationally at linear speeds which immediately before the fan entrance and at different radii from the fan axis are substantially equal to the linear speeds at the corresponding radii of the fluid in the fan.

2. A centrifugal compressor as claimed in claim 1, wherein the axes of adjustment are arranged radially about the fan axis.

3. A centrifugal compressor as claimed in claim 1 or 2, wherein the guiding elements comprise fixed blades whereof the outflow portion of each is angularly adjustable.

4. A centrifugal compressor as claimed in claim 3, wherein the fixed guide blades are curved helicoidally and receive the inflowing fluid from a helical conduit impressing on the fluid a rotational component corresponding to the mean average of the rotational speed component of the fluid in the fan.

5. A centrifugal compressor as claimed in claim 1 or 2, wherein the guiding elements comprise blades each angularly

adjustable so that the blades may be set to a desired direction of intake.

6. A centrifugal compressor as claimed in any of the preceding claims wherein the fan has straight-profiled vanes.

7. A centrifugal compressor as claimed in any of claims 1 to 5 wherein the fan has vanes whose inflow portion is curved to suit the direction of inflow in the direction of their motion.

8. A centrifugal compressor as claimed in any of the preceding claims, wherein the guiding elements consists in a number of orders of blades, the blades of one or more of which are angularly adjustable.

9. A centrifugal compressor as claimed in claim 8, wherein there are at least two orders of adjustable blades, and wherein the blades of one order are adjustable with angular motion to those of the other so as to obtain gradual directional change of flow.

10. A centrifugal compressor as claimed in claim 8 or 9, wherein the different orders of blades are adjustable either by a single controller or by means operating on the respective orders of blades separately.

11. A centrifugal compressor as claimed in any of the preceding claims, wherein the guiding elements comprise blades adjustable under the control of a gear wheel which is rotatable on a seat in the compressor's cowl by a lever actuating a pinion meshing with said gear wheel, the respective blades having pivots provided with pinions all meshing with and turnable simultaneously by said gear wheel.

12. A centrifugal compressor as claimed in any of the preceding claims in which the guiding elements comprise blades wholly or partly adjustable as regards their angular setting and wherein adjustment of the blades is effected by a servo-regulator controlled by the air-feed pressure required by the engine.

13. A centrifugal compressor substantially as hereinbefore described with reference to Figures 1 and 2, or Figures 3 and 4, or Figure 5, or Figures 6 and 7 of the accompanying drawings.

Dated this 4th day of July, 1940.

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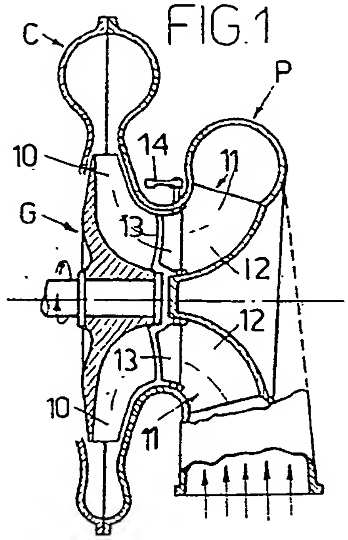


FIG.1

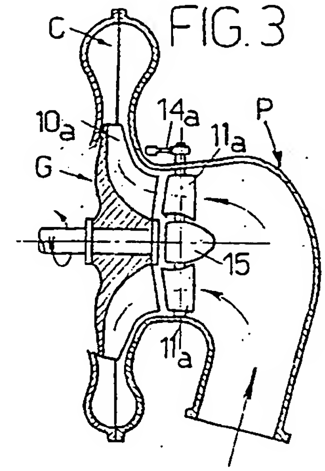


FIG.3

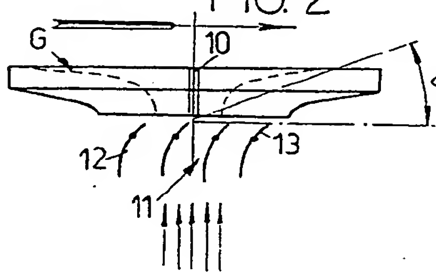


FIG.2

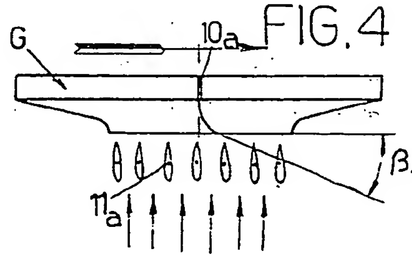


FIG.4

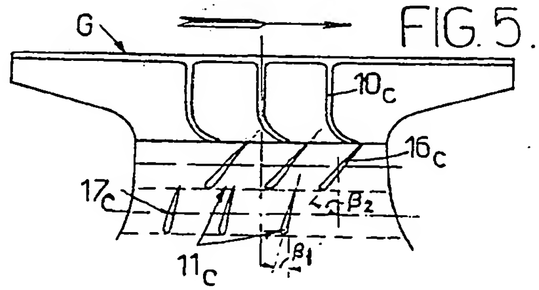


FIG.5.

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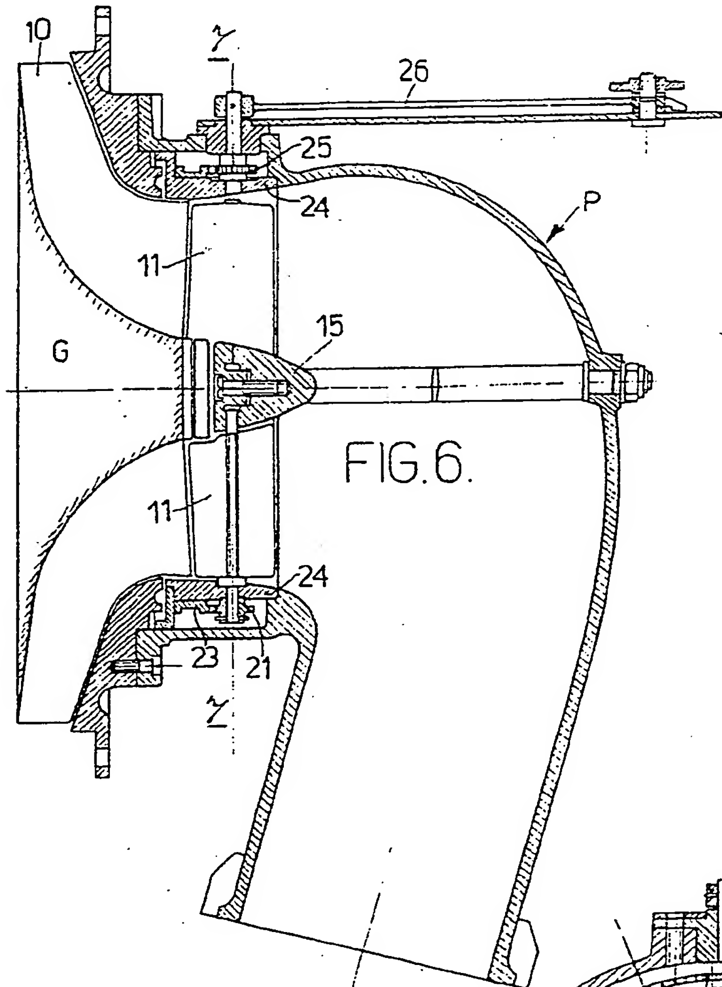



FIG.6.

FIG.7.

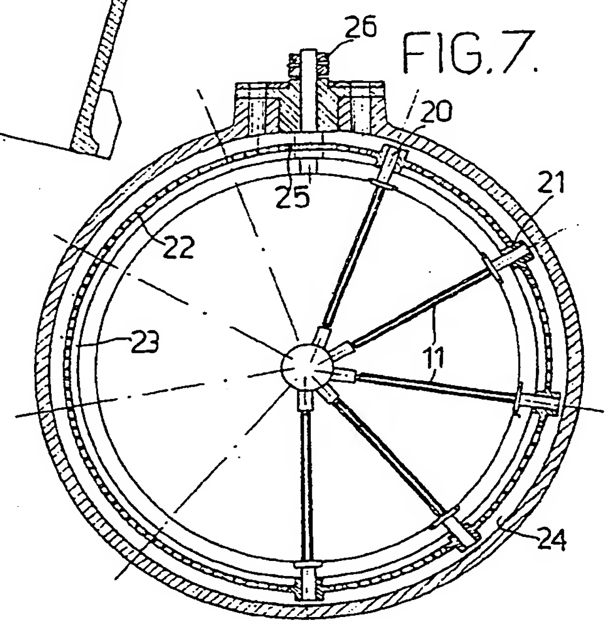
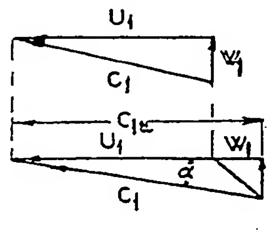



FIG.8.

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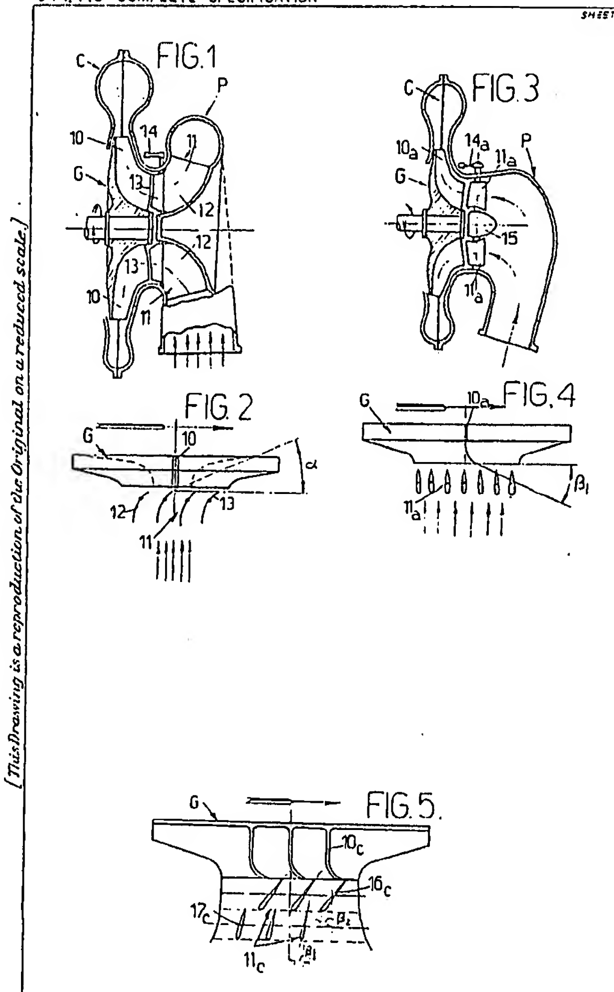


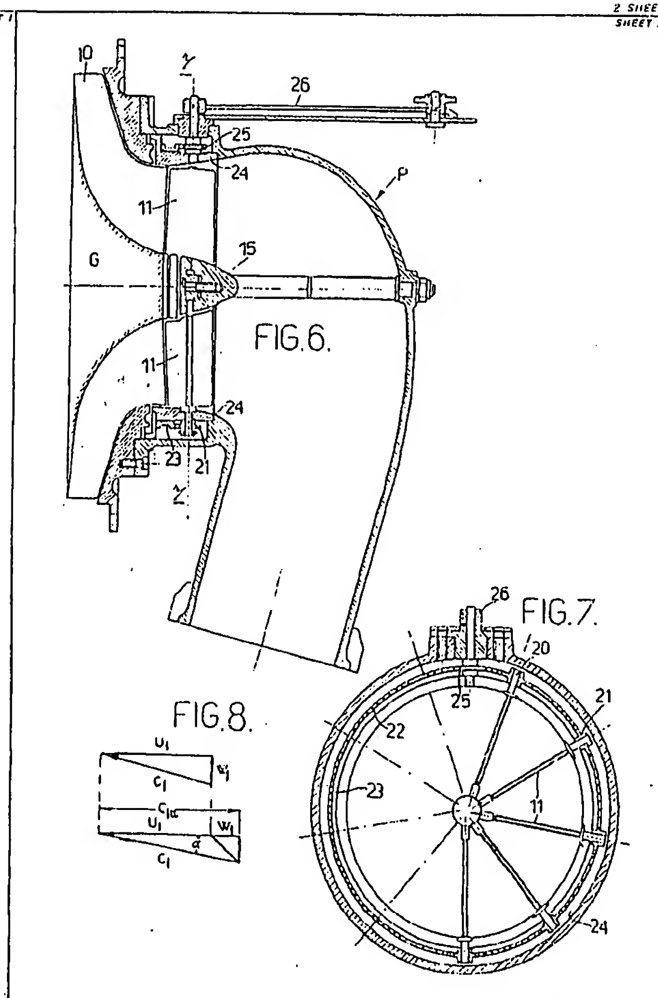


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